A METHOD OF IMPROVING THE PERFORMANCE
OF A LOW COMPRESSION ENGINE

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OF A LOW COMPRESSION ENGINE

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A METHOD OF IMPROVING THE PERFORMANCE OF A LOW COMPRESSION ENGINE

INTRODUCTION

This thesis is concerned with a proposed cycle for an internal combustion engine, including a thermodynamic analysis of the cycle and the results of applying the cycle to an actual engine.

The proposed cycle is a modification of the Otto cycle found so prevalently in automotive engines, and involves a use of different intake valve closing time to give a compression ratio different from the expansion ratio.

This idea was conceived by H. R. Ricardo, but no evidence was found that he tried the idea experimentally. Someone, however, did apply the cycle to a single-cylinder engine, without obtaining positive results.[1] The experimenter suggested that some advantage probably could be noted in a multicylinder engine, but again no evidence was found that an experiment had been carried out.

It, therefore, became the author's intention to apply the idea to a multicylinder engine to determine the characteristics of such an engine for comparison with engines of the usual Otto cycle type. It was his particular aim to determine if a more efficient engine might be obtained through use of the proposed cycle.

The cycle was ultimately applied to a 1937 Ford V-8 engine by

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[1]Arthur W. Gardiner and William E. Whedon, The Relative Performance Obtained with Several Methods of Control of an Overcompressed Engine Using Gasoline. N.A.C.A. Report No. 272.

altering the camshaft and cylinder heads and some useful data was obtained, which indicates the general characteristics and possibilities of an engine employing the proposed cycle. Considerable difficulty was encountered, however, in obtaining data when the altered heads cracked frequently and interrupted the work. Finally the engine threw a connecting rod, damaging the engine beyond repair, and bringing the work on this engine to an end.

At the time when the cylinder heads were giving constant trouble, it was decided to cease work on the V-8 engine and experiment with a single cylinder engine. It was realized that similar work had been done before, but it was strongly suspected that the reason no positive results were obtained was because of blow-back through the carbureter on the compression stroke, causing a loss of charge and a consequent decrease in efficiency. A remedy for this was felt to be in the use of a check valve between the carburetor and engine. This did prevent the loss of charge and brought about an increase in indicated efficiency, but cut down the brake horsepower by throttling to such an extent that no increase in brake efficiency was noted.

At this point it was decided to do some further work on the V-8 engine, in hopes of obtaining some more data that would better prove the thesis. The cylinder heads that had caused somuch difficulty had been altered by planing considerably and then adding brass to further reduce the clearance volume. This brazing, it was decided, had been responsible for the heads cracking later, because of stresses which had been set up in the structure. Therefore, some new heads were altered only by planing, no brass being added.

Some additional data was obtained, but very little, before the connecting rod broke and brought the experimental work on this engine to a permanent halt.

All of the work thus far mentioned was done in the Spring of 1947 and did not afford any positive proof of what performance could be expected from the design. Nor had sufficient analysis been made of the cycle to indicate the extent to which the idea should be applied, that is to say — how long after bottom center the intake valve should be left open and how much to correspondingly reduce the clearance volume.

Finally a method of analyzing the cycle was discovered in the Summer of 1950 and is presented below. This analysis shows clearly the possibilities of the cycle and indicates to what extent the idea should be applied for a particular engine of given characteristics.

Also it was decided that a single cylinder engine could be used to test the idea provided a surge tank was connected in front of a carburetor with an adjustable needle valve main jet. The surge tank would prevent loss of charge that was pushed back out the carburetor and the adjustable needle valve would allow adjustment of the air fuel ratio to prevent a rich mixture resulting from part of the charge going through the carburetor twice.

Most of the data and results presented in the following pages are the result of this last experiment. Some data was obtained from the V-8 engine which indicates the possibilities of the idea as applied to a multicylinder engine, but such data was not accurate enough to be presented as positive proof. It is presented, however, for what value it may be.

THERMODYNAMIC ANALYSIS

The proposed cycle will be applied to an internal combustion engine of the reciprocating four-stroke cycle type.

The conventional four-stroke cycle consists of intake, compression, power, and exhaust, each process corresponding to a single different stroke of the piston.

The proposed cycle also is a four-stroke cycle, but because of different intake-valve-closing time, the compression ratio and expansion ratio will not be equal, as is the case with the conventional cycle. The proposed cycle is shown on a PV diagram.

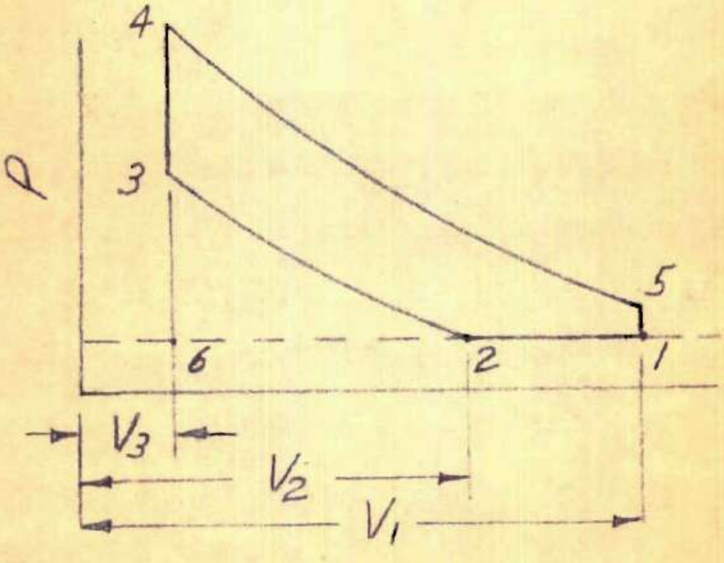

Figure 1. PV Diagram of Proposed Cycle

(a) From 6 to 1 the charge is drawn into the cylinder. In the actual engine this charge consists of an air and gasoline mixture. For this analysis it is assumed to be air only.

(b) From 1 to 2 the intake valve remains open so that compression occurs from 2 to 3 only. For analysis it is assumed atmospheric pressure will exist at points 1 and 2 and that the compression will be isentropic. At 2 the intake valve closes.

(c) From 3 to 4 heat is added at constant volume. In the actual engine this heat will come from burning of the charge itself. For this analysis it is assumed to be added to the air from an external source.

(d) Expansion takes place from 4 to 5, which is assumed to be isentropic for analysis.

(e) Heat is rejected from 5 to 1 at constant volume.

(f) The charge is expelled during the exhaust stroke 1 to 6.

The compression ratio, Rc, for this cycle is defined as $\frac{V2}{V3}$ and is limited in any practical engine by the characteristics of the fuel available.

$$(1) \quad Rc = \frac{V2}{V3}$$

The expansion ratio, R_E, is defined as $\frac{V1}{V3}$ and is largely the determining factor of the indicated efficiency of the engine.

$$(2) \quad R_E = \frac{V1}{V3}$$

For analysis let us consider an engine of given displacement so that the displacement, V_D, is constant and equal to C_1.

$$(3)\quad V_D = V_1 - V_3 = C_1$$

Also let us consider a fuel of given characteristics is to be used so that the compression ratio is limited and equal to a constant C_2.

$$(4)\quad R_C = \frac{V_2}{V_3} = C_2$$

Then let V_3, the clearance volume, vary as well as the closing time of the intake valve in order that V_2/V_3 will remain constant and equal to C_1 and consider the effects on the indicated efficiency of the engine; the brake power output; and the brake thermal efficiency of the engine.

The indicated efficiency E_I for the case of equal compression and expansion ratios may be calculated from a formula involving only the compression ratio. However, when the two ratios are different, as in this analysis, it becomes necessary to analyze each case separately.

Let us assume that we have an engine of one cubic foot displacement. Therefore, in equation (3) C_1 = 1 cu.ft. Also, let us limit our compression ratio to 6 so that in equation (4) $C_2 = 6$.

The medium for this analysis is air and we will assume that a full cylinder is drawn in on the intake stroke 6 to 1 and that at 1 the conditions will be 60°F. and 14.7 lbs/in^2 atmospheric pressure. These same conditions will exist on the compression stroke up to point 2 where the intake valve closes, it is assumed.

Compression from 2 to 3 will be assumed to be isentropic, as will the expansion from 4 to 5. The heat addition from 3 to 4 and the heat rejection from 5 to 1 are at constant volume. Intake and exhaust are assumed to require no work other than friction, which will not be considered on the indicated efficiency.

For air k = 1.4, the ratio of the specific heats, and C_v, the specific heat at constant volume, equals .1714. The gas constant R equals 53.35.

Indicated efficiencies will be determined for values of V_3 from 1/5 cu.ft., to 1/9 cu.ft., corresponding to expansion ratios of 6 and 10 to 1 respectively. The compression ratio V_2/V_3 will remain constant at 6. The heat added will be 100 BTU per cu.ft. of air.

Sample calculations are attached and the results obtained are as follows, for a compression ratio of 6 to 1:

TABLE I. Indicated Efficiencies of Proposed Cycle and Otto Cycle Having Equal Expansion Ratios

V_3	R_E	R_C Proposed	R_C Otto	E_I Proposed	E_I Otto
1/5	6	6	6	51.2	51.2
1/6	7	6	7	54.0	54.2
1/7	8	6	8	56.2	56.5
1/8	9	6	9	58.0	58.5
1/9	10	6	10	59.5	60.2

In the last column are the efficiencies obtained by using the formula for the efficiency of an Otto cycle having a compression ratio equal to the value given in the expansion ratio column.

It can be seen that this cycle then compares quite favorably with the Otto cycle for the same expansion ratios, as far as indicated efficiencies are concerned. The Otto cycle efficiencies are slightly higher, because with a higher compression ratio the heat added to the medium is added at a higher temperature.

The purpose of the analysis, however, is to determine the possibility of getting more brake efficiency out of a given size engine by means of the proposed cycle. The next step then is to show the comparative brake power outputs for the different expansion ratios.

The brake power output will depend upon the indicated efficiency, the volume of charge burned each cycle, the speed of the engine and the friction. For comparison the engine will be considered to run at some constant speed for which the friction, C_3, is considered constant, since the maximum pressure will be the same in every case. Another constant, C_4, dependent on the speed and heating value of the fuel is used to multiply by the volume of charge to give the heat supplied per unit of time.

In a practical engine the heat added will be from the burning of the charge drawn into the cylinder. The clearance volume will contain exhaust gases so that the volume of charge burned will equal $V_2 - V_3$ instead of V_2, which is the total volume of charge and burnt gases retained in the cylinder when the intake valve closes.

$$(5) \qquad V_c \text{ (volume of charge)} = V_2 - V_3$$

$$\text{but } V_2 = C_2 V_3 \text{ (from equation (4))}$$

$$(6) \qquad V_c = C_2 V_3 - V_3 = V_3(C_2 - 1)$$

$$(7) \qquad \text{Energy input} = V_c \times C_4 = C_4 V_3 (C_2 - 1)$$

$$\text{Power output} = \text{Energy input} \times \text{ind.eff.} - \text{friction power}$$

$$(8) \qquad \text{Power output} = C_4 V_3 (C_2 - 1)(E_I) - C_3$$

For this analysis, since we are considering an engine of given size, running at a constant speed, with a constant maximum pressure, it will be assumed that the power required to overcome friction is the same absolute value in every case. Further, this absolute value will be determined on the assumption that the mechanical efficiency of the engine is 80% where the compression ratio and expansion ratio are both 6. This is the case of greatest power output and therefore it is not likely for the friction to be greater in any of the other cases.

The brake thermal efficiency may now be determined. It is defined as output over input.

$$(9)\quad \text{Brake thermal efficiency} = \frac{C_4V_3(C_2-1)(E_I) - C_3}{C_4V_3(C_2-1)}$$

The following results were obtained from equations (8) and (9) for $C_2 = 6$, $C_3 = .20C_4V_3(C_2-1)E_I$ when $V_3 = 1/5$ and E_I = value from Table I on page 7. The results are for various values of V_3.

Table II. Efficiencies and Power Out-puts of Proposed Cycle for Different Expansion Ratios

V_3	E_I	Brake Power	Brake Efficiency	R_C	R_E	Indicated Power	Friction Power
1/5	.512	.410 C_4	.410	6	6	.512 C_4	.102 C_4
1/6	.540	.348 C_4	.418	6	7	.450 C_4	.102 C_4
1/7	.562	.300 C_4	.420	6	8	.402 C_4	.102 C_4
1/8	.580	.261 C_4	.417	6	9	.363 C_4	.102 C_4
1/9	.595	.229 C_4	.412	6	10	.331 C_4	.102 C_4

It has already been noted that the indicated efficiency for this cycle compares quite favorably with that of the Otto cycle for an equal expansion ratio. In fact, in the case of $V_3 = 1/5$ cu.ft., where the compression ratio and expansion ratio are the same and equal to 6, the proposed cycle is identical with the Otto cycle and of course, the indicated efficiency found is identical to that of the Otto cycle.

As the expansion ratio increases for the two cycles, the compression ratio is held to a limited value of 6 for the proposed cycle, whereas it increases and remains equal to the expansion ratio for the Otto cycle. This increase in compression ratio for the Otto cycle causes the Otto cycle to have an increasingly greater indicated efficiency than the proposed cycle at higher expansion ratios. It is noted, however, that this difference for an expansion ratio of 10 is only on the order of 1%.

The indicated power is found to decrease for the higher expansion ratios, because the clearance volume was decreased to give the higher expansion, and in order to keep the compression ratio constant, the amount of charge compressed was decreased by letting the intake valve close later. Thus less heat was added for the higher ratios, and in spite of a higher indicated efficiency, the power was found to decrease.

The power to overcome friction was given an assumed value, based on the assumption the engine has an 80% mechanical efficiency when operating on an Otto cycle of 6 to 1 compression ratio. The absolute value of power to overcome friction found from this calculation was assumed to be the value necessary in every other case.

The brake power was found by subtracting the friction power from the indicated power. Since the indicated power is less for the higher ratios, it is readily seen that the friction power will cut down the brake power a greater percentage for the higher expansion ratios, and result in a greater difference between indicated efficiency and brake efficiency.

Of course, it is logical to assume that for the higher expansion ratios, the power to overcome friction would decrease, since the higher pressures of compression and combustion would be maintained for a shorter period. There is no reason to believe the power to overcome friction would be greater for the higher expansion ratios. There is, however, a probability of a pumping loss from pushing part of the charge back out the carburetor. In the case of the single cylinder engine tested, this pumping loss was appreciable and showed up as an increase in friction when motoring the engine. In a multi-cylinder engine the loss should be much smaller because another cylinder on intake would have its intake work reduced by almost the amount of work done on the charge by the piston on compression. In such a case the loss would probably be negligible and is so assumed for this analysis.

It is seen that the brake power decreases with the increase of expansion ratio, even more rapidly than the indicated power. However, the brake efficiency is noted to increase to a maximum for an 8 to 1 expansion, and decrease from that point for greater expansion ratios. For different values of assumed friction, this maximum point would vary. For greater values it would reach a maximum at a smaller expansion ratio, whereas with smaller values it would reach a maximum at a greater ratio.

The point to be proven was that with a given size engine and a fuel of given characteristics, a higher brake efficiency could be attained with the proposed cycle than with the conventional Otto cycle.

The results of the analysis show that the proposed cycle gives a brake efficiency of .420 for an 8 to 1 expansion, whereas the Otto cycle for the same engine and fuel gives a brake efficiency of only .410.

This indicates very strongly the need for experimental work to substantiate the analysis. As pointed out before, it is expected that the mechanical efficiency of any given engine will largely determine the applicability of the proposed cycle. The higher the mechanical efficiency, the more the proposed cycle will increase the brake efficiency. And for too low a mechanical efficiency, the proposed cycle will do virtually no good.

To illustrate this point, calculations were made, using the procedure as before, for an engine of 75% mechanical efficiency and one of 85% mechanical efficiency. The brake efficiencies for different expansion ratios were as follows:

TABLE III. Brake Efficiency of Proposed Cycle for Different Expansion Ratios and Mechanical Efficiencies

R_E	V_3	Brake Efficiencies	
		75% E_M	85% E_M
6	1/5	38.4	43.5
7	1/6	38.6	44.7
8	1/7	38.4	45.5
9	1/8	37.6	45.8
10	1/9	36.6	45.8

These figures well illustrate the point in question. A much greater increase in brake efficiency for the higher values of R_E is noted for the 85% mechanical efficiency engine. In fact, for the 75% mechanical efficiency engine a decrease is noted for values of R_E greater than 8.

The analysis thus far shows the probability of an improvement in the efficiency of an engine at any given speed where the mechanical efficiency is high enough. However, it does notindicate what might be the effects on the overall operating characteristics of an engine to which the changes are made.

In order to see what these effects would probably be, let us take the characteristics of a known engine, such as the 1939 Chevrolet engine, and by means of certain assumptions make the changes and see what the effects are.

The characteristics of the engine are given in Mark's Handbook,[2] and the values as read from the curves are given in Table IV.

The RPM, Volumetric Efficiency, Brake Horsepower, Indicated Horsepower, and the lb-fuel/BHP-hr are all read directly from the curves.

The lb-fuel/IHP-hr was determined from the relation:

$$(10) \qquad \text{lb-fuel/IHP-hr} = \frac{\text{lb-fuel/BHP-hr} \times \text{BHP}}{\text{IHP}}$$

The Indicated Efficiency, E_I, was determined from the relation

$$(11) \qquad E_I = \frac{\text{IHP} \times 2545}{\text{lb/IHP-hr} \times 20{,}000}$$

The Mechanical Efficiency, E_M, was determined from the relation

$$(12) \qquad E_M = \frac{\text{BHP}}{\text{IHP}}$$

[2] Lionel S. Marks, Mechanical Engineers' Handbook, New York, McGraw-Hill Book Company, 1941, p. 1270, figure 9.

In order to simplify the analysis slightly, let us assume the compression ratio is 6 to 1 instead of 6.5 to 1, as it is given, so that the values given will correspond to $V_3 = 1/5$ in our analysis.

Then let us assume we are using a fuel of such octane rating that it is on the verge of knocking at the speed of maximum volumetric efficiency. The same fuel will be used for all changes, so that care must be taken in the analysis to insure that no increase in maximum compression will result.

Assuming that we have a displacement volume of 1, the volume of charge, V_c, instead of being as given in equation 6, which is for 100% volumetric efficiency, would be as follows:

$$(13) \quad V_c = V_3 (C_2 - 1) \times E_v \text{ (volumetric efficiency)}$$

Thus we are assuming that the intake valve is closing late enough in each case to insure that the maximum compression pressure is no greater than for the original engine.

For the speed of 800 RPM, where the maximum volumetric efficiency occurs for the original engine, equation (13) is used to calculate the volume of charge in each case. However, in the revised cases at other speeds the volume of charge remains constant as long as sufficient charge is being drawn into the cylinder, as indicated by the values for $V_3 = 1/5$. Excess charge is pushed back out on the compression stroke anyway for the revised cases. Where no excess charge is drawn in, none is pushed back out and the volume of charge for the revised case will equal that for the original engine. The volume of charge as determined in this manner is given in Table VI.

Following the same analysis procedure as before, the energy input is then determined from the following relation, C_4 being a constant involving the speed of the engine and heating value of the fuel.

(14) $$\text{Energy input} = V_c \times C_4$$

The Indicated Power equals the energy input times the Indicated Efficiency, E_I, so that

(15) $$\text{Indicated Power} = V_c \times C_4 \times E_I$$

The Brake Power output equals the Indicated Power minus Friction, C_3, so that

(16) $$\text{Brake Power} = (V_c \times C_4 \times E_I) - C_3$$

The Brake Thermal Efficiency equals output over input so that

(17) $$\text{Brake Thermal Efficiency } E_B = \frac{V_c \times C_4 \times E_I - C_3}{V_c \times C_4}$$

The Indicated Efficiency for each different value of V_3 is assumed to be in the same proportion to that of the original engine as are the respective indicated efficiencies given in the air standard analysis, Table II. Thus for any given speed

(18) $$\frac{E_I \text{ for } V_3 = 1/5}{E_I \text{ for } V_3 = X} = \frac{E_I \text{ for } V_3 = 1/5 \text{ Table II}}{E_I \text{ for } V_3 = X \quad \text{Table II}}$$

The Friction, C_3, in each case is determined from the Mechanical Efficiency, E_m, and Indicated Power of the original engine, $V_3 = 1/5$, at the particular speed.

(19) $$C_3 = V_c\, C_4\, E_I \times (100 - E_m)$$

The value for friction thus found for $V_3 = 1/5$ is used for every other value of V_3.

The Brake Thermal Efficiency may be evaluated from equation (17) without evaluating C_4 since it will cancel out after substituting for C_3 from (19). The Brake Efficiencies thus determined are given in Table VIII, with the corresponding values of lb-fuel/BHP-hr given in Table IX.

The Brake Power as evaluated from (16) is in terms of C_4, so that this constant must be evaluated from the known BHP for $V_3 = 1/5$ at each different speed, in order to express the output in terms of BHP. The values for BHP thus determined are given in Table VII.

The Torque output for each case may be evaluated from the relation

$$\text{(20)} \qquad \text{Torque (ft-lb)} = \frac{\text{BHP x 33,000}}{2\pi \text{ x RPM}}$$

The values of Torque corresponding to the values from Table VII were thus determined and are given in Table X.

The characteristics of the original engine and the four revisions are given in graphical form on page 2a. The values plotted are from Tables VI, VII, IX, and X.

TABLE IV. Characteristics of 1939 Chevrolet Engine

RPM	E_v	BHP	IHP	lb-fuel/BHP-hr	lb-fuel/IHP-hr	E_I	E_m
800	.83	24	27	.58	.515	.247	.890
1600	.80	49	58	.53	.448	.284	.845
2400	.80	71	88	.53	.428	.297	.805
3200	.73	78	105	.57	.423	.300	.742
3600	.67	75	109	.61	.420	.302	.688
4000	.60	68	108	.65	.410	.310	.630

TABLE V. Indicated Efficiencies from Equation (18)

V_3	Revolutions per Minute					
	800	1600	2400	3200	3600	4000
1/5	.247	.284	.297	.300	.302	.310
1/6	.260	.300	.313	.316	.318	.327
1/7	.271	.312	.326	.329	.331	.340
1/8	.280	.322	.336	.340	.342	.351
1/9	.287	.330	.345	.349	.351	.360

TABLE VI. Volume of Charge from Equation (13)

V_3	Revolutions per Minute					
	800	1600	2400	3200	3600	4000
1/5	.830	.800	.800	.730	.670	.600
1/6	.691	.691	.691	.691	.670	.600
1/7	.593	.593	.593	.593	.593	.593
1/8	.519	.519	.519	.519	.519	.519
1/9	.460	.460	.460	.460	.460	.460

TABLE VII. Brake Horsepower from Equation (16)

V_3	Revolutions per Minute 800	1600	2400	3200	3600	4000
1/5	24	49	71	78	75	68
1/6		43.9	63.0	77.5	81	73.7
1/7	18.2	38.3	54.3	66.5	71.8	77.2
1/8	16.1	33.7	47.2	57.5	61.5	65.6
1/9	14.4	29.9	41.6	49.1	53.5	56.4

TABLE VIII. Brake Thermal Efficiencies from Equation (17)

V_3	Revolutions per Minute 800	1600	2400	3200	3600	4000
1/5	.219	.240	.239	.223	.208	.195
1/6	.227	.249	.245	.234	.224	.212
1/7	.233	.253	.246	.234	.224	.224
1/8	.235	.254	.245	.232	.220	.218
1/9	.237	.254	.243	.228	.215	.210

TABLE IX. Pounds of Fuel per Brake Horsepower Hour Corresponding to Efficiencies in Table VIII

V_3	Revolutions per Minute 800	1600	2400	3200	3600	4000
1/5	.58	.53	.53	.57	.610	.652
1/6	.56	.510	.518	.543	.567	.600
1/7	.545	.503	.516	.543	.567	.568
1/8	.541	.501	.518	.548	.578	.583
1/9	.537	.501	.523	.557	.591	.605

TABLE X. Foot-pounds of Torque Corresponding to Vales of BHP Given in Table VII

V_3	Revolutions per Minute					
	800	1600	2400	3200	3600	4000
1/5	158	161	155	128	109	89.5
1/6	136	144	138	127	118	97.0
1/7	120	126	119	109	105	101
1/8	106	111	103	94.5	90.0	86.3
1/9	94.5	98.5	91.2	80.7	78.1	74.0

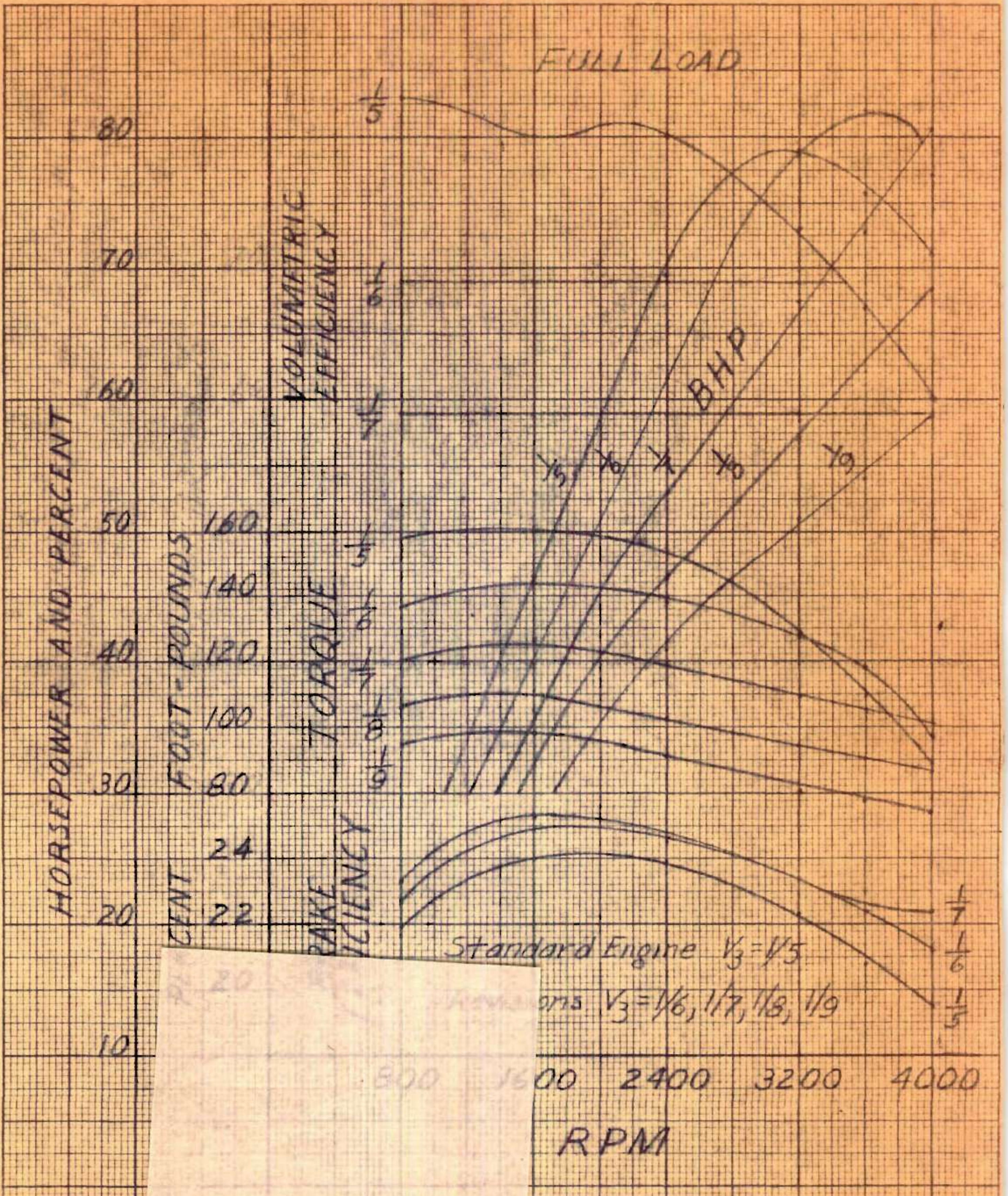

Figure 2. Characteristics of 1939 Chevrolet Engine and Four Revisions

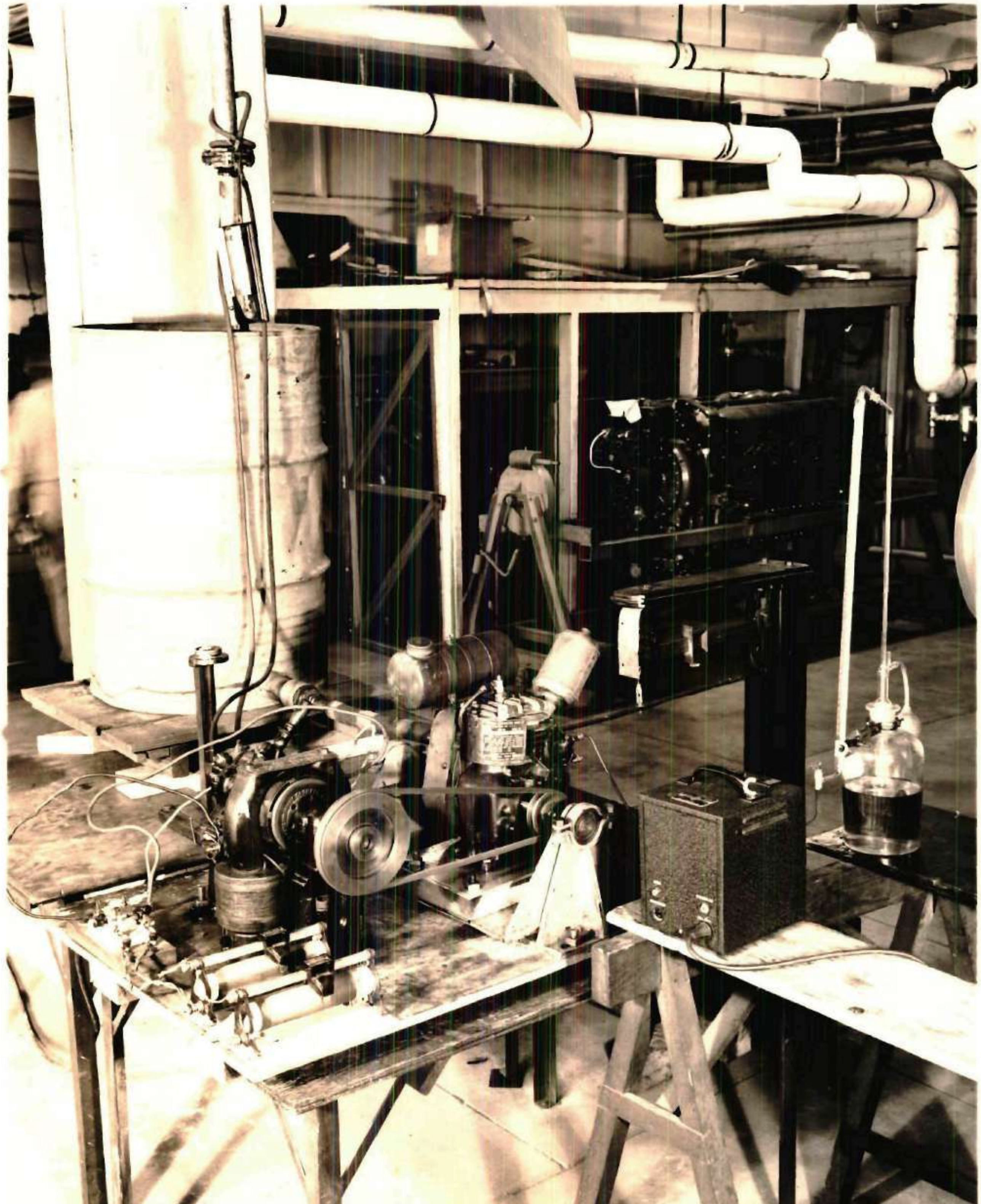

Figure A. Photograph of Apparatus - Front View

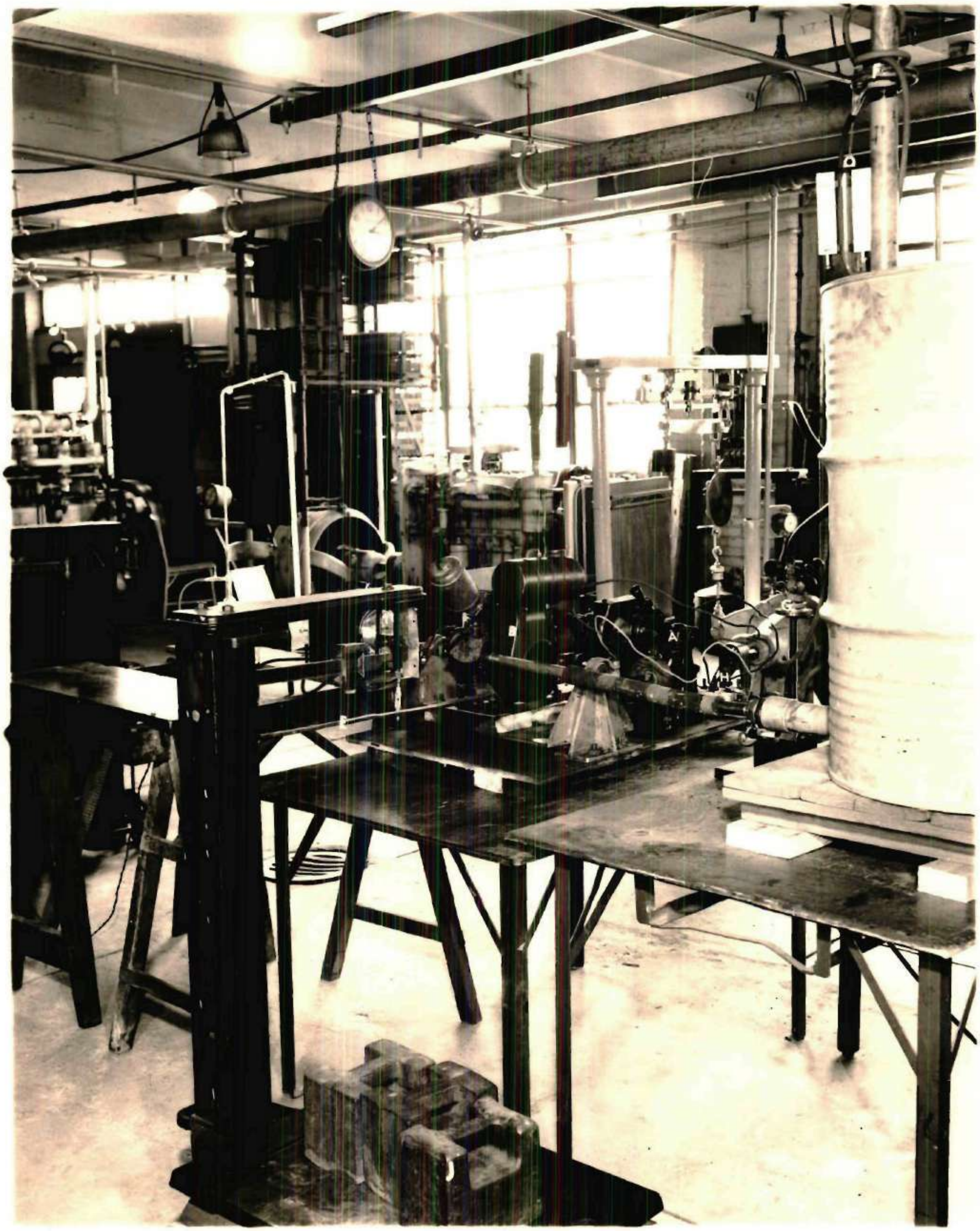

Figure B. Photograph of Apparatus - Rear View

APPARATUS AND TEST PROCEDURE

The apparatus used in testing the cycle proposed in this thesis is shown in the photographs on pages 21 and 22. It consists of the following items:

A. A Crocker Wheeler DC generator - motor rated at 1/2 HP or 4 amps at 110 volts. The speed rating is 1200 RPM, size 1/2 CM, No.2030, patent May 5, 1891.

B. A Briggs and Stratton Model NP 1-1/2 HP, four-stroke cycle, air cooled engine, - Bore = 2", stroke = 2", displacement = 6.28 in^3 or 103 cc.

C. A strobotac capable of measuring speeds from 600 RPM to 3700 RPM on the low scale and 2500 RPM to 14,500 RPM on the high scale. The manufacturer is General Radio Co., Cambridge, Mass., the type no. 631B, serial no. 12383.

D. A 50 cc graduate with 1/10 cc graduations for measuring the gasoline, combined with a half-gallon reservoir and burette.

E. Fairbanks scales with a 100 lb. slide balance scale having 1/2 lb. graduations. The ratio for balance between the platform and balance arm is 100 to 1.

F. An orifice measuring 17/32 of an inch diameter for measuring the air according to the formula $Q = .924\sqrt{w(h_1 - h_2)}$ where:

Q = air flow in pounds per minute

w = air density upstream in pounds per cubic foot

$h_1 - h_2$ = pressure drop across orifice in inches of alcohol

G. A micro-manometer connected across the orifice. It reads to .001 inch of alcohol from 0 to 8 inches.

H. A double pole, double throw switch rated at 15 amps and 125 volts.

J. Fifty pound standard weights for balancing the scales.

The wet and dry bulb thermometers may be seen hanging from the orifice connections.

A 55-gallon drum is used between the orifice and the engine to make the air flow smooth through the orifice as well as prevent loss of charge pushed back through the carburetor in the proposed cycle.

Two variable rheostats for varying the load may be seen in the first photograph, page 21, connected in parallel to one side of the double pole, double throw switch. They are rated at 110 volts and 2.5 amps each.

The rubber hose connection from the drum to the carburetor is clearly seen in the photograph on page 22. This connection is flexible enough to allow a balance to be reached on the scales. Care was taken to insure that the balance arm was made to point to precisely the same point on a marked piece of sheet metal each time a balance was effected. This procedure assured that the rubber hose and the parts of the cradle and engine were in exactly the same relation for each scale reading, thus practically eliminating errors due to friction or balance of the engine in the cradle.

The engine is mounted in a ball-bearing cradle, from which a 2-foot arm is extended and connected to the balance arm of the scales. The force exerted by the arm from the cradle is balanced by placing

50 pound standard weights on the scale platform with the sliding weight on the balance arm at zero. One or two excess weights are placed on the platform so that a fine adjustment may be made on the balance arm scale. The net reading equals the total weight on the platform minus the arm scale reading, all divided by 100, since that is the ratio of balance for these scales. The scales read to 1/2 pound, so that when used in reverse as in this case, the reading is to .005 pound.

The engine is connected to the motor-generator by means of a V-belt. The motor-generator serves as a starter and load for the engine as well as a means of motoring the engine to determine the friction.

The electrical connections are arranged as shown in Figure 3, so that by means of the double pole, double throw switch, the engine may be motored immediately after a run under load. By simply throwing the switch, the DC machine is changed from a generator with separate field excitation, to a D.C. shunt motor capable of motoring the engine. Thus the friction of the engine may be determined while the conditions are as near as possible equal to the load conditions.

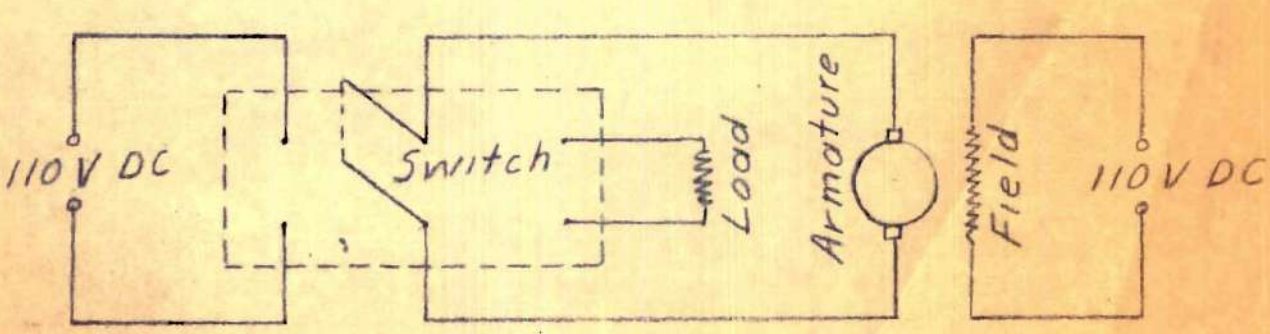

Figure 3. D.C. Machine Electrical Diagram

The fuel used was Amoco Ethyl having a specific gravity of 61° API, or .735, at 79°F. Sufficient gasoline was purchased at the beginning of the tests and kept in a closed container so that the same fuel with the same spedific weight could be used for all runs. The gasoline stayed at near enough a constant temperature, ±10°F, so that any change of specific weight due to temperature change was negligible. The heating value is assumed to be 20,000 BTU per pound for the calculation of thermal efficiency.

The fuel was supplied to the engine by means of the burette with 50 cc graduate pictured at D, on page 21. A stop watch was used on the efficiency tests to measure the time from a reading of 5 cc to a reading of 35 cc.

The output torque was measured with the scales at a lever arm distance of two feet. The output reading of the scale was corrected by the amount of the zero reading of the scale when the engine was idling with the belt disconnected.

The zero reading was large enough so that the friction torque could also be measured by noting the scale reading when the engine was being motored at the given speed. The friction scale reading was smaller than the zero reading but the difference in the two readings gives the correct results.

The clearance volumes for the standard and revised engines were determined by use of light oil to fill the clearance when the piston was at top dead center on compression. The quantity was measured with the same burette and graduate used for measuring the gasoline. Starting with the oil level at 0, it was admitted to the engine through the spark plug hole till the oil level reached the top of the plug hole. The reading of the graduate at this point gave the clearance volume in cc's, read to the nearest tenth directly.

The orifice used for measurement of the air was installed in a nominal 1-1/2" diameter pipe line. The inside diameter of the pipe is 1.61 inches and the diameter of the orifice is 17/32 inches or .531 inches.

The equation for air flow through an orifice takes the form of:

(1) $Q = C\sqrt{w(h_1 - h_2)}$

where Q = flow in pounds per minute

w = air density upstream in lb/ft^3

$h_1 - h_2$ = pressure drop across orifice in inches of alcohol

C = constant

This type equation holds for orifices constructed to standard specifications provided the Reynolds number for flow through the orifice is sufficiently high and provided the pressure drop is not over 2% of the upstream pressure.

The orifice used was constructed to standard specifications[3] and the pressure drop is well under 2%. However, the flow measured in the experiment is so small that the Reynolds number is below the specifications for using the coefficient of discharge given. The Reynolds number in this case is of the order of 10,000, whereas the lowest specified for use of the coefficient given is 30,000.

Referring to a graph given in Ower's book on page 118, it is shown that the coefficient of discharge for orifices of the type used in this

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[3]E. Ower, The Measurement of Air Flow. London, Chapman and Hall, 1949, p. 118.

experiment do vary with the Reynolds number, so that the coefficient for a Reynolds number of 10,000 would be some different than for a Reynolds number of 30,000.

However, it does seem evident from the graph that the coefficient of discharge for the orifice used would be very nearly constant for the range of Reynolds number encountered in this experiment, which would be from 5,000 to 15,000 at the extremes.

The coefficient given for orifices meeting the Reynolds number specification, as well as those met by the orifice used, is .599. Since the graph seems to indicate about a .01 greater coefficient for a Reynolds number of 10,000 than for 30,000 it was decided to use a coefficient of .61. This should provide sufficient accuracy for the purpose of this experiment, since the main objective is a comparison.

The equation derived by Ower for air flow through a standard orifice, as rearranged and simplified by A. W. Baker,[4] is as follows:

$$(2)\quad Q = 7.62\ aa' \sqrt{\frac{wr^2\ (h_1 - h_2)}{r^2 - 1}}$$

where Q = air flow in pounds per minute

a = orifice area in square inches

a' = coefficient of discharge

w = upstream air density in pounds per cubic foot

r = ratio of upstream pipe area to orifice area

$h_1 - h_2$ = pressure drop across orifice in inches of water

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[4]A. W. Baker, Jr. A Comparison of Various Vegetable Oils as Fuels for Compression Ignition Engines. A Georgia School of Technology thesis, 1946, p. 40.

By substituting the known quantities in equation (2) and converting so as to use the pressure drop reading in inches of alcohol, the following equation results:

$$(3)\quad Q = .924\sqrt{w(h_1 - h_2)}$$

where Q = air flow in pounds per minute

w = air density upstream in pounds per cubic foot

$h_1 - h_2$ = pressure drop across orifice in inches of alcohol - specific gravity of .790.

ENGINE ALTERATIONS

There were two revisions necessary to parts of the engine to give the proposed cycle. These alterations were made on the cylinder head and intake valve cam.

The cam was altered by adding brass to the closing side of the cam so as to cause the intake valve to remain open longer, fully closing only after the piston had completed approximately 50% of the compression stroke. Excess brass was added and the cam was then shaped by hand and the use of a lathe to give the desired results. A valve lift diagram, Figure 4 on page 31, illustrates the results obtained.

Tests were then run with this cam to determine what volumetric efficiencies would result, and these results were compared with the volumetric efficiencies obtained with the standard cam. It was found that the revised volumetric efficiencies were less, as was expected.

The maximum volumetric efficiencies in both cases were compared and it was determined that in the revised case the volumetric efficiency was approximately 15% less. On the basis of this information it was decided to reduce the clearance volume of the revised head 15% so that the maximum compression ratios would be again equal.

In order to do this, it was assumed that the two heads had the same clearance volume before alteration. The area was measured with a planimeter by tracing the irregular volume outline and found to be 6.5 sq. inches. It was desired to reduce the volume approximately 3 cc's and calculations showed that .028 inches planed off a standard head should accomplish this result.

A standard head was accordingly cut down on a lathe .028 inches and it was assumed that the head thus altered had a volume 3 cc's less than another standard head left unaltered. However, later measurements with a graduate reading to 1/10 cc showed the difference in the two heads to be only 1.9 cc, as shown on page 32.

There is little doubt that the .028 inches planed off reduced the volume of that particular head exactly 3 cc's. The only logical explanation is that the head which was altered had 1 cc greater volume to start with than the head retained as standard.

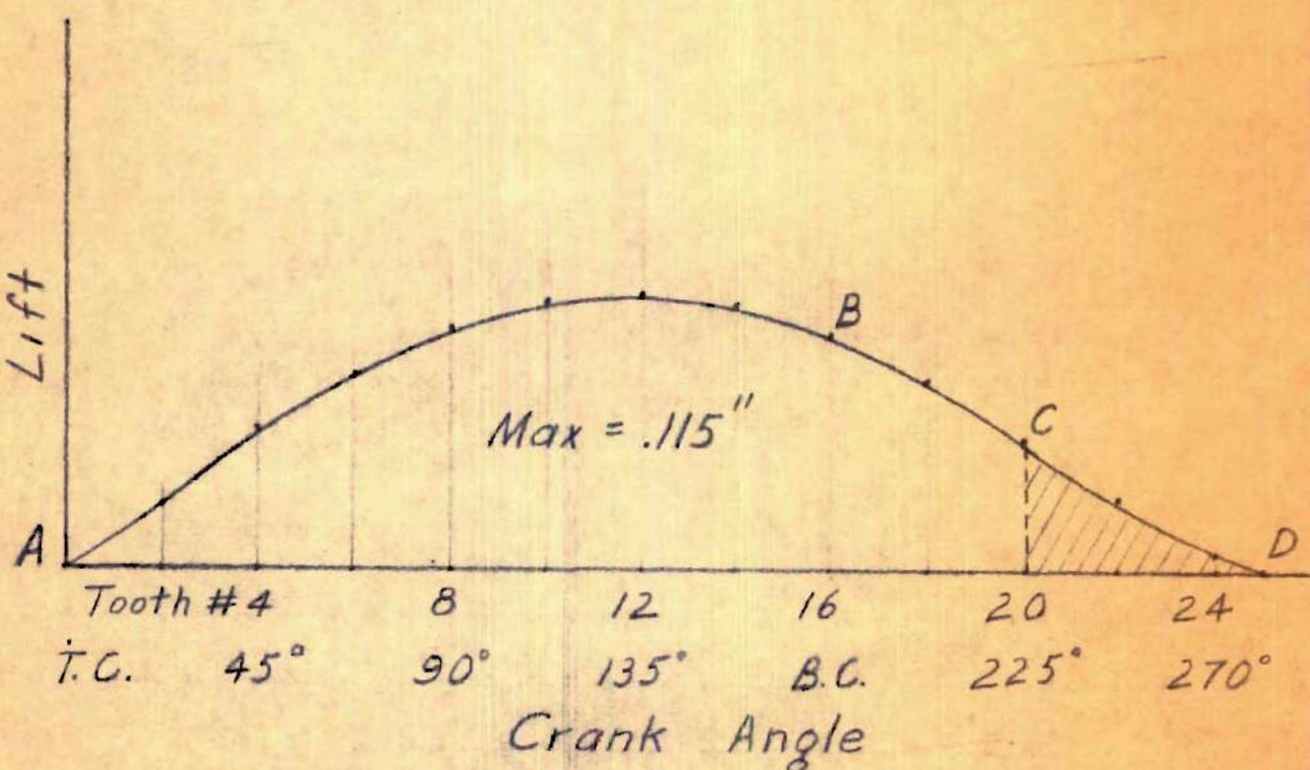

Figure 4. Intake Valve Lift Diagram

EXPERIMENTAL RESULTS

Clearance Volumes

Standard Head = 23.2 cc

Revised Head = 21.3 cc

Displacement Volume (both cases) = 103 cc

Expansion Ratios

$$\text{Standard Head} = \frac{103 + 23.2}{23.2} = 5.45$$

$$\text{Revised Head} = \frac{103 + 21.3}{21.3} = 5.84$$

TABLE XI. Standard Engine Characteristics
(Standard Head and Standard Cam)

Speed RPM	Volumetric Efficiency %	Actual Compression Ratio	Torque Output ft-lb	Brake Horsepower H.P.
2150	69.6	4.09	3.02	1.24
2410	73.4	4.26	3.24	1.49
2650	72.2	4.21	3.12	1.57
2880	72.0	4.20	3.00	1.65
3100	72.2	4.21	2.90	1.71
3350	68.0	4.01	2.74	1.75
3525	64.9	3.88	2.54	1.71
3700	63.9	3.84	2.64	1.86
3900	65.5	3.91	2.56	1.90
4025	64.6	3.86	2.26	1.73
4275	63.0	3.80	2.18	1.77
4400	61.6	3.74	2.12	1.77
4550	60.4	3.68	1.96	1.70

TABLE XII. Revised Engine Characteristics
(Revised Head and Revised Cam)

Speed RPM	Volumetric Efficiency %	Actual Compression Ratio	Torque Output ft-lb	Brake Horsepower H.P.
1800	52.4	3.53	2.00	.685
1960	56.9	3.75	2.22	.83
2010	55.4	3.68	2.34	.90
2080	54.8	3.65	2.34	.93
2535	61.6	3.98	2.70	1.31
2645	60.9	3.94	2.54	1.28
2710	60.9	3.94	2.50	1.29
3140	65.3	4.16	2.38	1.42
3430	65.7	4.18	2.44	1.59
3700	62.0	4.00	2.44	1.72
3975	61.0	3.95	2.20	1.67
4350	61.6	3.98	2.06	1.71
4550	62.2	4.01	1.96	1.70

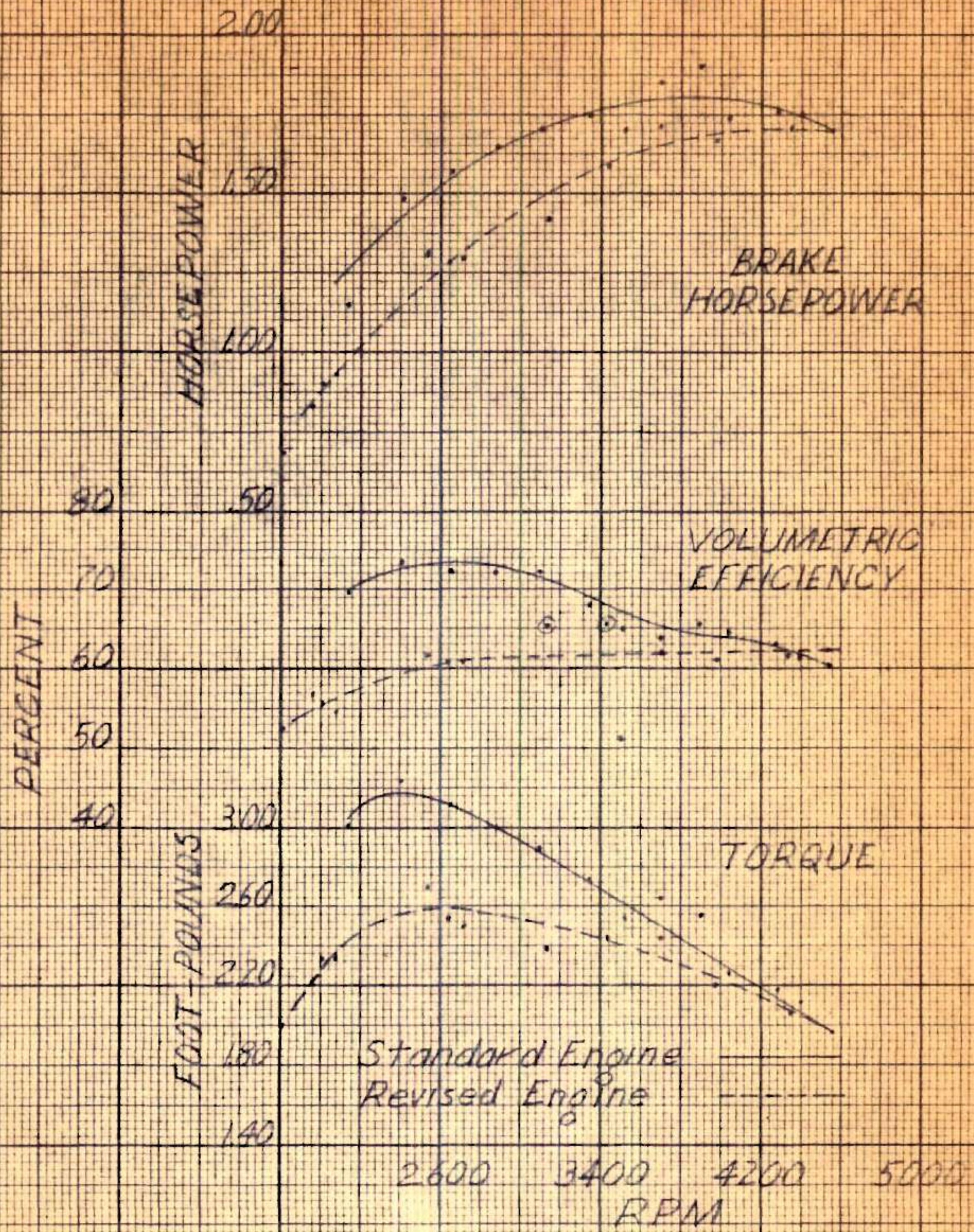

Figure 5. FULL LOAD CHARACTERISTICS OF BRIGGS & STRATTON MODEL NP ENGINE AND ONE REVISION

TABLE XIII. Efficiency Tests on Engine With Standard Head and Standard Cam

Item							
1	Run No.	1	2	3	4	5	6
2	Speed RPM	2375	2390	2660	2675	4660	4667
3	Torque ft-lb	2.93	2.93	3.14	3.16	1.90	1.90
4	Brake Horsepower H.P.	1.32	1.33	1.59	1.61	1.69	1.69
5	Fuel lb/hr	.982	.955	1.32	1.35	1.34	1.465
6	Fuel lb/BHP-hr	.741	.715	.830	.838	.795	.865
7	Fuel lb/IHP-hr	.621	.601	.691	.703	-	-
8	Brake Thermal Efficiency %	17.2	17.8	15.3	15.2	16.0	14.7
9	Indicated Thermal Efficiency %	20.5	21.2	18.4	18.1	-	-
10	Volumetric Efficiency	71.1	70.5	74.0	74.0	58.1	58.2
11	Air-fuel Ratio	13.4	13.75	11.61	11.41	15.65	14.38
12	Friction Torque ft-lb	.58	.58	.62	.62	-	-
13	Friction H.P.	.262	.264	.316	.314	-	-
14	Indicated H.P.	1.58	1.59	1.91	1.92	-	-
15	Air lb/hr	13.15	13.12	15.32	15.40	20.95	21.05
16	Mechanical Efficiency %	83.5	83.6	83.3	83.8	-	-

TABLE XIV. Efficiency Tests on Engine With Revised Head and Revised Cam

Item						
1 Run No.	1	2	3	4	5	6
2 Speed RPM	2370	2370	2664	2660	4600	4650
3 Torque ft-lb	2.48	2.48	2.50	2.50	1.86	1.86
4 Brake Horsepower H.P.	1.12	1.12	1.27	1.27	1.63	1.61
5 Fuel lb/hr	.835	.830	1.078	1.065	1.353	1.426
6 Fuel lb/BHP-hr	.747	.741	.848	.841	.830	.884
7 Fuel lb/IHP-hr	.564	.561	.649	.642	-	-
8 Brake Thermal Efficiency %	17.1	17.2	15.0	15.1	15.4	14.4
9 Indicated Thermal Efficiency %	22.6	22.7	19.6	19.8	-	-
10 Volumetric Efficiency %	60.9	60.9	62.0	61.4	61.9	61.2
11 Air-fuel Ratio	13.42	13.51	11.68	11.66	16.03	15.2
12 Friction Torque ft-lb	.80	.80	.76	.76	-	-
13 Friction H.P.	.361	.361	.386	.385	-	-
14 Indicated H.P.	1.48	1.48	1.66	1.66	-	-
15 Air lb/hr	11.21	11.21	12.6	12.42	21.7	21.7
16 Mechanical Efficiency %	75.7	75.7	76.5	76.5	-	-

TABLE XV. Efficiency Tests on Engine With
Standard Head and Revised Cam

Item				
1	Run No.	1	2	3
2	Speed RPM	2340	2340	2350
3	Torque ft-lb	2.42	2.42	2.42
4	Brake Horsepower H.P.	1.080	1.080	1.083
5	Fuel lb/hr	.820	.852	.835
6	Fuel lb/BHP-hr	.760	.790	.770
7	Fuel lb/IHP-hr	.585	.608	.595
8	Brake Thermal Efficiency %	16.75	16.12	16.55
9	Indicated Thermal Efficiency %	21.8	20.9	21.4
10	Volumetric Efficiency %	61.5	61.5	61.7
11	Air-fuel Ratio	13.85	13.35	13.73
12	Friction Torque ft-lb	.72	.72	.72
13	Friction H.P.	.321	.321	.321
14	Indicated H.P.	1.401	1.401	1.404
15	Air lb/hr	11.36	11.36	11.46
16	Mechanical Efficiency %	77.0	77.0	77.0

TABLE XVI. Output Characteristics of 1937 Ford V8 Engine with Standard Parts. Full Throttle

RPM	Dynamometer lb	Torque Ft-lb	Horsepower
2200	160	140	58.7
2300	157	137	60.2
2400	155	136	62.0
2550	152	133	64.6
2600	151	132	65.5
2700	148	129	66.5
2900	140	122	67.6
3150	136	119	71.4

Note: Dynamometer torque arm = 10.5 inches

TABLE XVII. Output Characteristics of 1937 Ford V8 Engine with Revised Camshaft and Standard Heads Full Throttle

RPM	Dynamometer lb	Torque Ft-lb	Horsepower
1950	120	105.0	39.0
2200	120	105.0	44.0
2400	116	101.5	46.4
2700	111	97.0	50.0
3200	110	96.5	58.6

TABLE XVIII. Output Characteristics of 1937 Ford V8 Engine with Revised Camshaft and Heads. Expansion Ratio = 7.8 to 1. Full Throttle

RPM	Dynamometer lb	Torque Ft-lb	Horsepower
2000	125	109	41.6
2200	123	108	45.0
2400	123	108	49.2
2600	124.5	109	54.0
3200	120	105	64.0

TABLE XIX. Output Characteristics of 1937 Ford V8 Engine with Revised Camshaft and Heads. Expansion Ratio = 11 to 1. Full Throttle

RPM	Dynamometer lb	Torque Ft-lb	Horsepower
1900	130	114	41.2
2100	132.5	116	46.5
2400	135.4	118	54.2
2700	133.8	117	60.0
3000	130	114	65.0
3600	115.5	101	69.3

TABLE XX. Efficiency Tests of 1937 Ford V8 Engine
with Revised Camshaft and Standard Heads
Full Throttle

Run	RPM	lb-Fuel/Hr	Horsepower	lb-Fuel/Hp-hr
1	2000	23.6	38.3	.615
2	2000	23.6	38.0	.620
3	2200	26.0	41.8	.620
4	2200	25.3	41.8	.605
5	2200	25.3	42.0	.602
6	2400	29.4	47.4	.620
7	2750	32.8	51.7	.635

TABLE XXI. Efficiency Tests of 1937 Ford V8 Engine
with Revised Camshaft and Heads
Expansion Ratio = 7.8 to 1. Full Throttle

Run	RPM	lb-Fuel/Hr	Horsepower	lb-Fuel/HP-hr
1	2000	23.8	39.0	.610
2	2025	23.8	39.7	.599
3	2225	24.7	44.5	.555
4	2200	24.1	44.4	.543
5	2350	28.7	47.5	.604
6	2500	30.4	51.7	.588
7	2600	30.9	53.7	.575

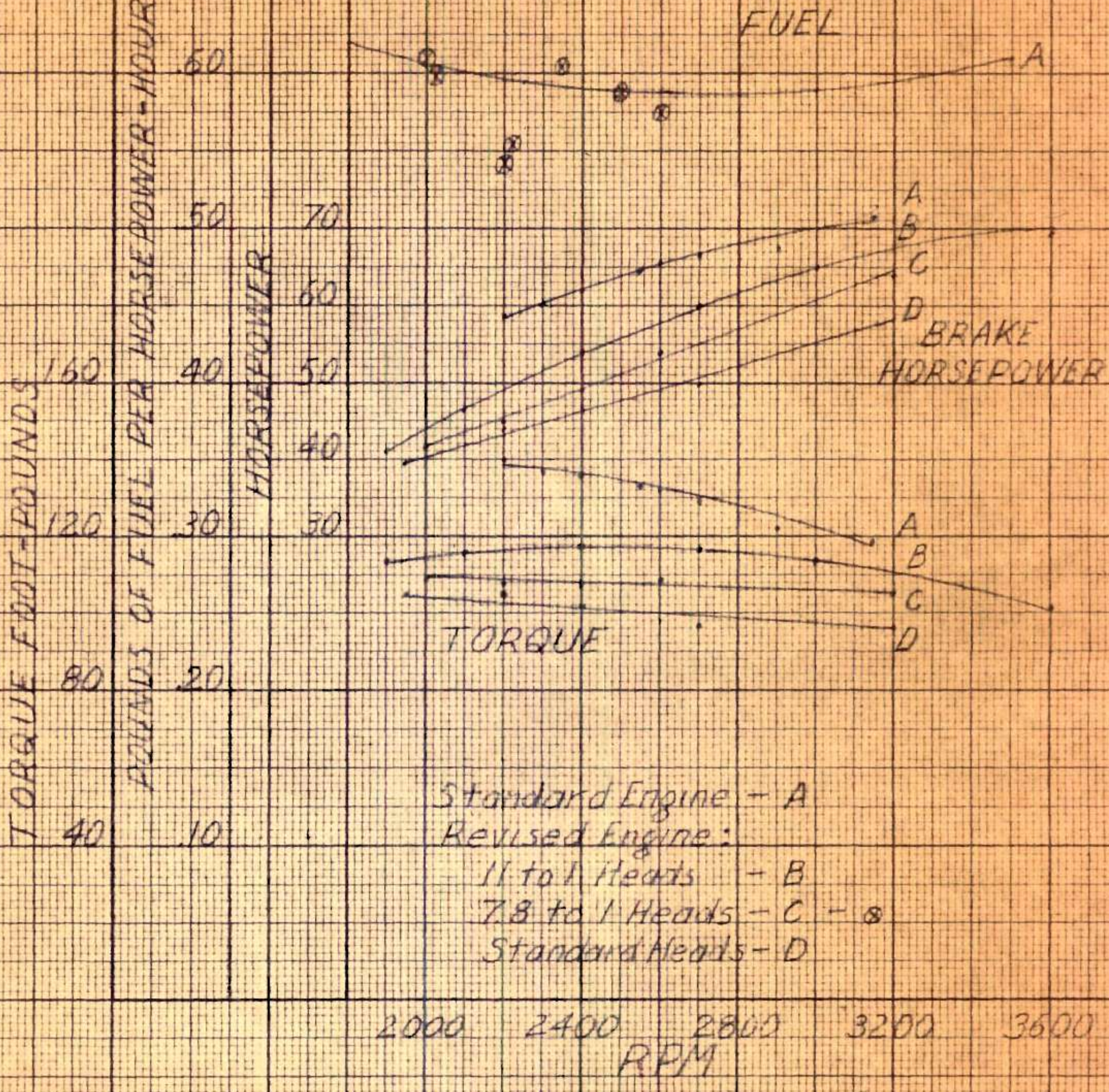

Figure 6. FULL LOAD CHARACTERISTICS OF 1937 FORD V8 ENGINE AND THREE REVISIONS

DISCUSSION

A basic assumption was made in analyzing the proposed cycle, as well as applying it to an engine, that volumetric efficiency is largely the determining factor of knock in a particular engine where the mechanical design is already fixed. For instance, in a given engine operating on fuel somewhat below the required octane rating for the engine, knocking will be most pronounced at speeds where the volumetric efficiency is highest, for normal carburetion and spark timing corresponding to the speed.

Consequently in the normal Otto cycle gasoline engine having a volumetric efficiency curve which peaks considerably, the compression ratio is limited by this one peak. In order to prevent excessive, damaging knock for these peak conditions, the compression ratio must be lower than that required for other speeds.

In the proposed cycle it is assumed that the proper intake valve closing time will give a volumetric efficiency curve which practically is constant through the entire operating speed range. This assumption is shown graphically in Figure 2, page 20. For the original engine, $V_3 = 1/5$, the volumetric efficiency curve drops off sharply beyond a speed of 2400 RPM. In the two revisions shown, $V_3 = 1/6$ and $1/7$, the late closing of the intake valve causes a decrease in the volumetric efficiency at low speeds because of part of the charge actually being pushed back out of the cylinder on compression. This is assumed to cause a constant volumetric efficiency curve up to the speed where the curve intersects the curve for the standard engine.

Actually, the volumetric efficiency of the revised engine was found to remain practically constant even beyond this intersection point, as may be seen in Figure 5, page 35. This probably is because the late closing time of the intake valve allows more charge to come into the cylinder and only a small part is pushed back out at the very high speeds.

If we calculate the volume of charge retained in the cylinder each cycle, add to this the clearance volume and divide the result by the clearance volume, we get what we shall call actual compression ratio. The volume of charge equals the displacement volume times the volumetric efficiency at a given speed.

In the analysis made, it is assumed that this actual compression ratio is the determining factor in a particular engine of the anti-knock quality needed in the fuel. This assumes normal carburetion and spark timing for the different speeds.

Referring to Table XI - "Standard Engine Characteristics," for a speed of 2410 RPM, the actual compression ratio is 4.26, and for a speed of 4550, the ratio is 3.68. Assuming proper carburetion and a spark advance normal for the particular speed, a fuel with considerably lower octane rating could be used for the lower ratio.

In other words, it is assumed that where the actual compression ratios are equal for two different speeds, the octane requirements will be the same for practical purposes.

Referring to Table XII, "Revised Engine Characteristics," for a speed of 2535, the actual compression ratio is 3.98 and we find the same value for a speed of 4350. The octane requirements for these two different speeds should be equal according to the assumption.

Referring to Figure 5, the volumetric efficiency curve is seen to be practically constant at speeds above 2600, except for two points at

3140 and 3430 RPM. These two points are circled and were not considered in drawing the average curve because it is strongly suspected they are the result of a tuned effect in the intake air system at that particular speed range. The air enters the carburetor through a 40" straight section, and this could very conceivably cause a tuned effect at these speeds that would affect the quantity of charge being pushed back out the cylinder. Probably what happens is that a higher pressure results in the intake manifold at the point in the cycle when charge is expelled from the cylinder. This higher pressure than usual causes less charge to be expelled, leaving more in the cylinder, and a higher volumetric efficiency results.

If this is true, no such points would result with a short intake system where the air enters the carburetor directly. The curves for Torque in Figure 6 for a V8 engine, where the air entered directly, show no peculiar points corresponding to any tuned effect, indicating that the above conclusions are sound.

Therefore, in an engine with a flat volumetric efficiency curve such as the dotted curve in Figure 5, the clearance volume can be reduced from that of the original design by the percentage difference in the peaks of the volumetric efficiency curves, this without altering significantly the fuel anti-knock requirements.

In this case the peak of the standard engine curve is about 73.5 and the peak of the revised engine curve is 62.2. The percentage reduction here is $\frac{73.5 - 62.2}{73.5} \times 100 = 15.4\%$. The original clearance volume given on page 32 is 23.2 cc's, and for the revised case this could be reduced $23.2 \times .154 = 3.6$ cc's. This would give a clearance volume of

$23.2 - 3.6 = 19.6$ cc's. The actual clearance volume used for the revised case, page 32, was 21.3 cc's. For a clearance volume of 19.6 cc, the expansion ratio would be $\frac{103 - 19.6}{19.6} = 6.25$ and the actual compression ratio peak vaolue $\frac{(.622 \times 103) - 19.6}{19.6} = 4.26$.

For the original engine the expansion ratio is 5.45, page 32, and the peak value of actual compression ratio $\frac{(.735 \times 103) - 23.2}{23.2} = 4.26$.

With the clearance volume actually used in the revised engine the expansion ratio is 5.84, page 32, and the peak value of actual compression ratio is $\frac{(.622 \times 103) - 21.3}{21.3} = 4.00$.

On this basis of defining actual compression ratio and assuming that it is the determining factor for octane requirements, it is seen that the revised engine octane requirements are lower for the clearance volume used than for the original engine, and that the clearance volume could be further reduced from 21.3 cc to 19.6 cc to equalize the octane requirements. This would increase the expansion ratio as defined by equation (2), page 5, from 5.84 to 6.25 as compared to an expansion ratio of 5.45 for the original engine.

It is interesting to note the efficiencies of the standard and revised engines in light of the facts just stated. Although the clearance volume was not reduced the full amount that the analysis shows it could have been, the experimental results show a definite increase in indicated thermal efficiency for the revised engine. The results are shown in Tables XIII, XIV, AND XV.

Note from Table XV for the revised cam and standard head that the indicated thermal efficiency for a speed of 2340 to 2350 averages about 21.4%. Note from Table XIV that the average indicated thermal efficiency for a speed of 2370 is 22.65. Then note from Table XIII for a speed of 2375 to 2390 that the efficiency averages 20.85 for the standard engine.

If it is agreed that the speeds involved are close enough for comparison, note the marked increase in indicated efficiency for the revised engine -- 22.65% compared to 20.85% for the standard engine. This is an increase of $\frac{22.65 - 20.85}{20.85} = 8.62\%$, considerably more than would be expected based on an air standard analysis of the expansion ratios involved.

It should be noticed also that the revised engine with the standard head shows a higher indicated efficiency (21.4) than the standard engine (20.85). A possible explanation is that the revised cycle involves less charge being compressed, which would mean a lower maximum temperature and less heat loss.

It will be noticed from these tables that the brake thermal efficiencies are essentially equal for comparable speeds for the revised and standard engines. This fact at first seems hard to explain in view of the marked difference in indicated efficiencies. The answer lies in comparison of friction horsepower for the two cases.

Notice that for the standard engine the friction is considerably less than for the revised engine at comparable speeds. This increase in friction for the revised engine is probably due to a pumping loss and not mechanical friction. In the proposed cycle some charge is pushed back out the carburetor on compression, and in a single cylinder engine the work done to push it out is a complete loss. Also this action reverses the flow in the intake manifold and increases the work of intake when the next cycle begins. Thus in a single cylinder engine there is a double loss due to the expelling action on compression and this can very well account for the pumping loss which shows up as an increase in friction when motoring the engine.

However, in a multicylinder engine this pumping loss might well be negligible - at least it should be much less than for the single cylinder. When the charge is expelled from one cylinder on compression there is another cylinder on intake ready to take the excess. Consequently the work done on the charge is not by any means a complete loss since the work of intake is reduced, probably by almost the amount of work done on the charge.

Some data was obtained from a multicylinder engine - a 1937 Ford V8 - which tends to substantiate this theory, although no actual measurement of friction was made. However, the fact that a sizeable increase in brake efficiency was obtained, as may be seen in Figure 6, is some indication that the friction was not much different for the two cases.

It can be shown from the data obtained for the single cylinder engine that if the friction were the same for both cases, that the brake efficiency would show up greater for the revised case than for the standard engine. In other words, if we take the indicated torque as measured for the revised engine, subtract the friction torque as measured for the standard engine, and calculate the brake horsepower of the revised engine from this, we will find the revised engine has the highest brake efficiency, based on the fuel consumption as measured for the revised engine.

In line with this, note that the method used in measuring engine output and friction inherently gives a more accurate comparison of indicated values than for brake or friction values. The engine and motor-generator were belt-connected with the engine mounted in a cradle so that engine torque reaction was directly measured, independent of belt slip. Thus the reaction was measured with the engine at full load, then with the engine being motored at the same speed by the D.C. machine immediately following the run. The difference in the two readings should give an accurate indication of indicated torque and horsepower for comparison, since any errors due to method should be practically identical for both cases.

The weak point in the measurement is the determination of a zero point to determine what part of the measured values is friction and what part brake output. The method decided upon was to disconnect the belt and let the engine idle. Only by idling the engine would the scales show reasonable sensitivity because of the rubber hose connection to the engine. The belt was disconnected because even at idling, some reaction would result from overcoming friction and windage in the D.C. generator. By this method a reading was obtained which is felt to be accurate to plus or minus .02 pounds.

However, determination of the zero point has no bearing whatsoever on measurement of indicated output, and for this reason comparison of the indicated values is felt to be the most accurate.

It is interesting to compare the curves in Figures 2, 5 and 6 for similarity in what was predicted by analysis in Figure 2 and what was found by experiment in Figures 5 and 6.

Note first the similarity in volumetric efficiency curves in Figures 2 and 6. For the revised case in both instances the volumetric efficiency is constant through a wide range and at high speeds is found to be equal to the standard engine value. A dissimilarity exists at very low speeds, however. In the analysis we assumed a perfectly flat curve for even the lowest speeds. However, experiment shows in Figure 6 that the volumetric efficiency drops off at low speeds.

This can be explained from the valve closing charcteristics shown in Figure 4. Charge is drawn into the cylinder on the intake stroke from A to B and charge actually continues to enter the cylinder after the piston starts up, to some point C. At point C the pressure in the cylinder becomes equal to the manifold pressure and beyond C charge is pushed back into the manifold until the valve is fully closed at D.

Two things probably happen at low speed to cause lower volumetric efficiency. Point C will be much nearer point B leaving more valve opening and time for charge to be expelled. Also at low speeds, charge can be expelled through small valve openings with much less pressure drop than at high speeds. These combined effects cause a lower pressure to exist at point D for low speeds, indicating less volumetric efficiency.

Note from this analysis also the probability that at very high speeds intake will continue practically to point D and thus cause a higher volumetric efficiency for the revised case than for the standard engine. This is substantiated by experiment as shown in Figure 5 where the revised curve is seen to cross the standard engine curve at 4550 RPM.

Comparing Figures 2, 5 and 6 further, note the similarities in Torque and Brake horsepower curves. In the analysis as well as results, the torque for the revised case is seen to be lower at low speeds, but at some high speed the torque values are equal, beyond which the revised engine value should be greater. This last point was not proved experimentally but is logical to assume from the trend of the curves.

The Brake horsepower curves are similar in this same respect - low for the revised engine at low speeds but approaching each other and becoming equal at some high speed, both from analysis and experiment.

The analysis shows the probability of obtaining a greater maximum horsepower for the right revision than for the standard engine. This was not proved experimentally, probably because full advantage was not taken to reduce the clearance volume the full amount allowable in the revised case. The extra efficiency which would be obtained by making the further reduction could very well result in a brake horsepower increase sufficient to surpass the maximum value for the standard engine.

CONCLUSIONS

The experimental results presented seem to bear out in close detail the results obtained by analysis and indicate strongly that a more efficient multi-cylinder engine with a higher maximum horsepower can be built by employing the proper application of the proposed cycle, without requiring any higher octane rating of fuel.

Also, analysis can be made to show that the application of the cycle is equally beneficial regardless of the fuel and compression ratio already employed in a standard Otto cycle engine. In other words, an engine already using a 12 to 1 compression ratio and fuel accordingly, can be improved as much with respect to efficiency and maximum brake horsepower as one using a compression ratio of 4 to 1 and fuel accordingly, the revised engine to use the same fuel as the standard engine in the given case.

The only disadvantage to the proposed cycle seems to be a sacrifice of power at low speeds, which is shown in Figures 2, 5 and 6. How serious this drawback may be, of course, depends on the particular application. It would seem, though, that it could largely be overcome with the proper use of gears or torque converters.

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APPENDIX I. DATA

Table XXII Standard Engine Data

Speed RPM	Manometer Reading " Alcohol	Scale Reading Pounds	Date
2150	.620	3.00-.29	11/27/50
2410	.865	3.00-.18	"
2650	1.008	3.00-.24	"
2710	1.052	3.00-.30	"
2880	1.190	3.00-.30	"
3100	1.385	3.00-.35	"
3350	1.430	3.00-.43	"
3525	1.444	3.00-.53	"
3700	1.541	3.00-.48	"
3900	1.805	3.00-52	11/28/50
4025	1.875	3.00-.67	"
4275	2.010	3.00-.71	"
4400	2.036	3.00-.74	"
4550	2.091	3.00-.82	"
4660	2.022	3.00-.86	"

11/27/50 Barometer = 28.81" Hg
DB = 73°F
WB = 57°F
Air density = .07130 lb/ft^3

11/28/50 Barometer = 29.01" Hg
DB = 80°F
WB = 63°F
Air density = .07080 lb/ft^3

Scale zero reading = 1.50-.30 = 1.20 lb

TABLE XXIII. Revised Engine Data

Speed RPM	Manometer Reading " Alcohol	Scale Reading Pounds	Date
1800	.244	3.00-.70	12/11/50
1960	.341	3.00-.59	"
2010	.339	3.00-.53	"
2080	.356	3.00-.53	"
2535	.670	3.00-.35	"
2645	.712	3.00-.43	"
2710	.748	3.00-.45	"
3140	1.155	3.00-.51	"
3430	1.394	3.00-.48	"
3700	1.438	3.00-.48	"
3975	1.611	3.00-.60	"
4350	1.976	3.00-.67	"
4550	2.220	3.00-.72	"

Barometer = 28.84" Hg
DB = 79°F
WB = 63°F
Air density = .0705 lb/ft^3

Scale zero reading = 1.50-.20 = 1.30 lb

TABLE XXIV. Efficiency Test Data on Standard Engine

Time min-sec	Graduate Reading cc	Speed RPM	Micro Manometer "Alcohol	Scale Reading Pounds	Ambient Temperature DB°F	 WB°F
			Run No. 1			
0	5	2370	.783	3-.565	77	60
1	15	2375	.783	3-.565	78	61
2	25.5	2375	.783	3-.565	78	61
2'58"	35	2380	.783	3-.565	78	61
Av. 2'58"	30 cc	2375	.783	2.435	78	61
			Run No. 2			
0	5	2385	.774	3-.565	78	61
1	14.8	2400	.780	3-.565	78	61
2	24.8	2390	.780	3-.565	78	61
3'3"	35	2385	.780	3-.565	78	61
Av. 3'3"	30 cc	2390	.778	2.435	78	61

Date 11/17/50 Barometer = 29.14" Hg

Scale zero = 1.50-.53 = .97 lb

Friction scale reading @ 2380 RPM = 1.00-.32 = .68 lb.

Air density = .07140 lb/ft^3

TABLE XXV. Efficiency Test Data on Standard Engine

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	$DB^{o}F$	$WB^{o}F$
			Run No. 3			
0	5	2660	1.060	3-.24	73	57
1	17.8	2660	1.060	3-.24	73	57
2	31	2660	1.060	3-.24	73	57
2'18"	35	2660	1.060	3-.24	73	57
Av. 2'18"	30 cc	2660	1.060	2.76	73	57
			Run #4			
0	5	2690	1.092	3-.24	73	57
1	19	2675	1.080	3-.24	73	57
2	33	2675	1.080	3-.24	73	57
2'9½"	35	2660	1.070	3-.24	73	57
Av. 2'9½"	30 cc	2675	1.081	2.76	73	57

Date 11/27/50 Barometer = 28.81" Hg

Scale zero = 1.50-.32 = 1.18 lb

Friction zero reading @ 2660 RPM = 1.50-.63 = .87 lb

Air density = .07130 lb/ft^3

TABLE XXVI. Efficiency Test Data on Standard Engine

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	DB^oF	WB^oF
			Run No. 5			
0	7	4660	2.022	3-.86	82	64
1	21.5	4660	2.022	3-.86	82	64
2	35	4660	2.022	3-.86	82	64
2'10"	37	4660	2.022	3-.86	82	64
Av. 2'10"	30 cc	4660	2.022	2.14	82	64
			Run No. 6			
0	17	4675	2.069	3-.86	82	64
1	32.3	4665	2.045	3-.86	82	64
1'20"	37	4660	2.022	3-.86	82	64
Av. 1'20"	20 cc	4667	2.045	2.14	82	64

Date 11/28/50 Barometer = 29.01" Hg

Scale zero = 1.50-.31 = 1.19 lb

Air density = .07080 lb/ft^3

TABLE XXVII. Efficiency Test Data on Revised Engine

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	DB^{o}F	WB^{o}F
			Run No. 1			
0	3	2370	.575	2.5-.31	80	68
1	11.5	2370	.575	2.5-.31	80	68
2	19.3	2370	.575	2.5-.31	80	68
3	28.6	2370	.575	2.5-.31	80	68
3'29"	33	2370	.575	2.5-.31	80	68
Av. 3'29"	30 cc	2370	.575	2.19	80	68
			Run No. 2			
0	2	2370	.575	2.5-.31	80	68
1	10.8	2370	.575	2.5-.31	80	68
2	18.7	2370	.575	2.5-.31	80	68
3	27.5	2370	.575	2.5-.31	80	68
3'30"	32	2370	.575	2.5-.31	80	68
Av. 3'30	30 cc	2370	.575	2.19	80	68

Date 11/16/50 Barometer = 29.20 " Hg

Scale Zero = 1.50-.55 = .95 lb

Friction scale reading @ 2370 RPM = 1.50-.95 = .55 lb

Air density = .07115 lb/ft^3

TABLE XXVIII. Efficiency Test Data on Revised Engine

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	DB°F	WB°F
			Run No. 3			
0	5	2675	.750	3-.52	89	70
1	16	2660	.737	3-.52	89	70
2	26.6	2660	.737	3-.52	89	70
2'42"	35	2660	.737	3-.52	89	70
Av. 2'42"	30 cc	2664	.740	2.48	89	70
			Run No. 4			
0	5	2660	.720	3-.52	90	71
1	15.7	2660	.720	3-.52	90	71
2	27	2660	.720	3-.52	90	71
2'44"	35	2660	.720	3-.52	90	71
Av. 2'44"	30 cc	2660	.720	2.48	90	71

Date 12/1/50 Barometer = 29.08" Hg

Scale zero = 1.50-.27 = 1.23lb

Friction scale reading @ 2660 RPM = 1.50-.65 = .85 lb

Air density = .06975 lb/ft^3

TABLE XXIX. Efficiency Test Data on Revised Engine

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	DB°F	WB°F
			Run No. 5			
0	5	4600	2.200	3-.84	90	70
1	20	4600	2.200	3-.84	90	70
2	33	4600	2.200	3-.84	90	70
2'9"	35	4600	2.200	3-.84	90	70
Av. 2'9"	30 cc	4600	2.200	2.16	90	70
			Run No. 6			
0	5	4650	2.200	3-.84	90	70
1	20.7	4650	2.200	3-.84	90	70
2	34	4650	2.200	3-.84	90	70
2'5"	35	4650	2.200	3-.84	90	70
Av.2'5"	30 cc	4650	2.200	2.16	90	70

Date 12/1/50 Barometer = 29.08" Hg

Scale zero = 1.50-.27 = 1.23 lb

Air density = .06975 lb/ft^3

TABLE XXX. Efficiency Test Data on Engine With Standard Head and Revised Cam

Time	Graduate Reading	Speed	Micro Manometer	Scale Reading	Ambient Temperature	
min-sec	cc	RPM	" Alcohol	Pounds	DB°F	WB°F
			Run No. 1			
0	5					
1	13.6	2340	.580	2.5-.29	72	57
2	22.5	2340	.580	2.5-.29	72	57
3	31.3	2340	.580	2.5-.29	72	57
3'33"	35	2340	.580	2.5-.29	72	57
Av. 3'33"	30 cc	2340	.580	2.21	72	57
			Run No. 2			
0	5	2340	.580	2.5-.29	72	57
1	13.5	2340	.580	2.5-.29	72	57
2	22.3	2340	.580	2.5-.29	72	57
3	30.9	2340	.580	2.5-.29	72	57
3'25"	35	2340	.580	2.5-.29	72	57
Av. 3'25"	30 cc	2340	.580	2.21	72	57
			Run No. 3			
0	6	2350	.590	2.5-.29	72	57
1	14.3	2350	.590	2.5-.29	72	57
2	22.3	2350	.590	2.5-.29	72	57
3	31.4	2350	.590	2.5-.29	72	57
3'29"	36	2350	.590	2.5-.29	72	57
Av. 3'29"	30 cc	2350	.590	2.21	72	57

Date 11/17/50 Barometer = 29.14" Hg
Air density = .0723 lb/ft^3
Scale zero = 1.50-.50 = 1.00 lb
Friction reading = 1.00-.36 = .64 lb

TABLE XXXI. Intake Valve Lift Data

Tooth No.	Crank Angle Degrees	Micrometer Reading Inches	Lift Inches
0-32	Top Center	.609	0.000
2	22.5	.581	0.028
4	45	.549	0.060
6	67.5	.528	0.081
8	90	.510	0.099
10	112.5	.498	0.111
12	135	.494	0.115
14	157.5	.498	0.111
16	180	.513	0.096
18	202.5	.530	0.079
20	225	.556	0.053
22	247.5	.581	0.028
24	270	.601	0.008
26	292.5	.609	0.000
28	315	.609	0.000
30	337.5	.609	0.000
32-0	360 T.C.	.609	0.000

TABLE XXXII. Efficiency Tests of 1937 Ford V8 Engine with Revised Camshaft and Standard Heads Full Throttle

Run	Dynamometer lb	RPM	Time/lb-Fuel
1	115	2000	2' 32.5"
2	114	2000	2' 32.8"
3	114	2200	2' 18.8"
4	114	2200	2' 22.2"
5	114.5	2200	2' 22.0"
6	118.5	2400	2' 2.3"
7	112.5	2750	1' 49.8"

TABLE XXXIII. Efficiency Tests of 1937 Ford V8 Engine With Revised Camshaft and Heads. Expansion Ratio - 7.8 to 1. Full Throttle

Run	Dynamometer lb	RPM	Time/lb-Fuel
1	117	2000	2' 31.4"
2	117.3	2025	2' 31.2"
3	120	2225	2' 25.8"
4	121	2200	2' 29.4"
5	121	2350	2' 5.4"
6	124	2500	1' 58.3"
7	124	2600	1' 56.7"

APPENDIX II

SAMPLE CALCULATIONS

Table I.

Refer to Figure 1, p.4

1. $V_3 = 1/6$ for example

2. V_1 = displacement volume + V_3

$= 1 \text{ cu.ft.} + 1/6 = 1\ 1/6 \text{ ft}^3 = V_5$

3. $C_2 = 6$ given

4. $V_2 = C_2V_3 = 6V_3 = 6 \times 1/6 = 1 \text{ ft}^3$

5. $P_1 = P_2 = 14.7 \text{ lb/in}^2$ given

6. $T_1 = T_2 = 60^oF + 460 = 520^oR$ given

7. $v_1 = v_2 = \frac{RT}{P} = \frac{(53.35)(520)}{(14.7)(144)} = 13.1 \text{ ft}^3/\text{lb}$

8. $P_3 = P_2 \left|\frac{V_2}{V_3}\right|^k = 14.7\left(\frac{1}{1/6}\right)^{1.4} = 180 \text{ lb/in}^2$

9. $T_3 = T_2\left(\frac{V_2}{V_3}\right)^{k-1} = 520\left(\frac{1}{1/6}\right)^{.4} = 1065^oR$

10. Add 100 BTU per cu.ft. of air compressed

11. ${}_3Q_4 = 100 \times V_2 = 100 \times 1 = 100 \text{ BTU}$

12. $T_4 - T_3 = \frac{{}_3Q_4}{C_vM} = \frac{100}{.1714\ \frac{1}{13.1}} = 7640^oR$

13. $T_4 = 1065 + 7640 = 8705^oR$

14. $P_4 = P_3\frac{T_4}{T_3} = 180\ \frac{8705}{1065} = 1472 \text{ lb/in}^2$

15. ${}_2W_3 = \frac{144}{k-1}\ p_2V_2\left[1-\left(\frac{V_2}{V_3}\right)^{k-1}\right] = \frac{144}{1.4-1}\ (14.7)(1)\left|1-6^{.4}\right| = 5540 \text{ ft-lb}$

16. ${}_4W_5 = \frac{144}{k-1}\ p_4V_4\left[1-\left(\frac{V_4}{V_5}\right)^{k-1}\right] = \frac{144}{.4}\ (1472)(1/6)\left[1-\left(\frac{1}{7}\right)^{.4}\right] = 47{,}900 \text{ ft-lb}$

17. ${}_1W_2 = 144(p_1V_1 - p_2V_2) = 144\ p_1(V_1 - V_2) = 144(14.7)(1\ 1/6 - 1) = 353 \text{ ft-lb}$

18. Net work $= {}_4W_5 - {}_2W_3 - {}_1W_2 = 47{,}900 - 5540 - 353 = 42{,}007$ ft-lb

19. $E_I = \frac{42{,}007}{100 \times 778.2} = 54.0\%$

Table II.

1. $V_3 = 1/6$ cu.ft. for example

2. $E_I = .540$ from Table I.

3. $C_2 = 6$

4. $C_3 = .20C_4V_3(C_2 - 1)E_I$ when $V_3 = 1/5$, and $E_I = .512$

   $C_3 = .1024C_4$

5. Brake power $= C_4V_3(C_2 - 1)(E_I) - C_3$ from Equation (8), p.9

   Brake power $= C_4(1/6)(6-1)(.540) - .102C_4 = .348C_4$

6. Brake Efficiency $= \dfrac{C_4V_3(C_2 - 1)(E_I) - C_3}{C_4V_3(C_2 - 1)}$ from Equation (9), p.10

   Brake Efficiency $= \dfrac{C_4(1/6)(6-1)(.540) - .102C_4}{C_4(1/6)(6-1)} = .418$

7. R_C = Compression Ratio = 6 by statement of problem

8. R_E = Expansion Ratio $= \dfrac{V_1}{V_3} = \dfrac{1\ 1/6}{1/6} = 7$

9. Friction Power $= C_3 = .102C_4$

10. Indicated Power = Brake Power + Friction Power

$$= .348C_4 + .102C_4 = .450C_4$$

Table III.

Calculations same as for Table II.

Table IV.

1. RPM = 800 Read from curve.[5]
2. E_v = volumetric efficiency = 83% curve
3. BHP = 24 H.P. curve
4. IHP = 27 H.P. curve
5. lb/BHP-hr = .58 curve
6. lb/IHP-hr = Item 5 x $\frac{\text{Item 3}}{\text{Item 4}}$ = .58 x 24/27 = .515
7. E_I = Indicated thermal efficiency = $\frac{2545}{(6)\text{x}20{,}000} = \frac{2545}{.515\text{x}20{,}000}$ = .247
8. E_m = mechanical efficiency = (3)/(4) = $\frac{24}{27}$ = .890

Table V.

1. V_3 = 1/7 for example
2. RPM = 3200 for example
3. E_I for $V_3 = 1/7$ = $\underset{\text{(Table IV)}}{E_I \text{ for } V_3 = 1/5}$ x $\frac{E_I \text{ for } V_3 = 1/7 \text{ Table II}}{E_I \text{ for } V_3 = 1/5 \text{ Table II}}$

$$= .300 \times \frac{.562}{.512} = .329$$

Table VI.

1. V_3 = 1/6 for example
2. RPM = 3600 for example
3. V_c, volume of charge, = $V_3(C_2-1) \times E_v$ = 1/6(6-1).83 = .691

   However, where V_c for V_3 = 1/5 is less, this value is taken so

   V_c = .670

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[5]Lionel S. Marks, Mechanical Engineers' Handbook, McGraw-Hill Book Co. Inc., 1941, p. 1270, figure 9.

Table VII.

1. $V_3 = 1/7$ for example

2. RPM = 3200 for example

3. Brake horsepower $= V_c C_4 E_I - C_3$ Equation (16)

$V_c = .593$ Table VI

$E_I = .329$ Table V

$C_3 = V_c C_4 E_I \times (100 - E_m)$ for $V_3 = 1/5$ Equation (19)

$C_3 = .730C_4 .300 \times (100 - .742) = .0565C_4$

Brake horsepower $= (.593)(C_4)(.329) - .0565C_4 =$

$.195C_4 - .057C_4 = .138C_4$

C_4 is evaluated from known brake horsepower

for $V_3 = 1/5$ @ 3200 RPM

$78\text{BHP} = .730C_4 .300 - .057C_4 = .162C_4$

$$C_4 = \frac{78}{.162} = 481$$

Brake horsepower $= .138C_4 = .138 \times 481 = 66.5$ H.P.

Table VIII

1. $V_3 = 1/7$ for example

2. RPM = 3200 for example

3. Brake thermal efficiency $= \dfrac{V_c C_4 E_I - C_3}{V_c C_4}$ -Equation (17)

$$= \frac{.593C_4 .329 - .057C_4}{.593C_4} = .234$$

Table IX.

1. $V_3 = 1/7$ for example

2. RPM = 3200 for example

3. $\text{lb-fuel/BHP-hr} = \dfrac{2545}{20{,}000 \times E_B} = \dfrac{2545}{20{,}000 \times .234} = .543 \text{ lb/HP-hr}$

Table X.

1. $V_3 = 1/7$ for example

2. RPM = 3200 for example

3. $\text{Torque} = \dfrac{\text{BHP} \times 33{,}000}{2 \quad \times \text{RPM}} = \dfrac{66.5 \times 33{,}000}{2 \quad \times 3{,}200} = 109 \text{ ft-lb}$

Table XI. Data in Table XXII.

1. RPM = 3100 for example

2. Scale reading = 3.00 - .35 lb data

   = 2.65 lb

3. Zero scale reading = 1.20 lb data

4. Lever arm = 2.0 ft data

5. Torque = (2.65 - 1.20) x 2.0 = 2.90 ft-lb

6. $\text{Brake horsepower} = \dfrac{2 \quad \times \text{Torque} \times \text{RPM}}{33{,}000} = \dfrac{2 \quad \times 2.90 \times 3100}{33{,}000} = 1.71 \text{ H.P.}$

7. $\text{Volumetric efficiency} = \dfrac{\text{ft}^3 \text{ air/min}}{\text{ft}^3 \text{displacement/min}} = \dfrac{Q}{D}$

$$Q = .924 \sqrt{w(h_1 - h_2)} \text{ lb/min} \quad \text{equation (3) p.29}$$

$$Q = \frac{.924}{w} \sqrt{w(h_1 - h_2)} \text{ ft}^3/\text{min} = .924 \sqrt{\frac{h_1 - h_2}{w}} \text{ ft}^3/\text{min}$$

$$D = \frac{6.28 \text{ in}^3}{1728} \times \frac{\text{RPM}}{2} = \frac{\text{RPM}}{550} \text{ ft}^3/\text{min suction displacement}$$

Table XI cont'd.

7. cont'd.

$$\text{Volumetric efficiency} = \frac{Q}{D} = \frac{.924\sqrt{\frac{h_1 - h_2}{w}}}{\frac{RPM}{550}} = \frac{508}{RPM}\sqrt{\frac{h_1 - h_2}{w}}$$

$$= \frac{508}{3100}\sqrt{\frac{1.385}{.07080}} = 72.2\%$$

$h_1 - h_2 = 1.385''$ Alcohol data

$w = .07080\ lb/ft^3$ data Table XXII read from chart in M.E. laboratory

for given dry bulb, wet bulb and barometer readings.

8. Actual compression ratio $= \frac{E_v \times V_D + \text{clearance volume}}{\text{clearance volume}}$

$$= \frac{(.722 \times 103) + 23.2}{23.2} = 4.21$$

Table XII. Data in Table XXIII

Calculations same as for Table XI.

Table XIII. Data in Tables XXIV, XXV, XXVI.

1. Run No.1 for example

2. Speed = 2375 RPM data Table XXIV

3. Torque = (Scale reading - zero reading) x 2 Table XXIV

$= (3 - .565 - .97) \times 2 = 2.93$ ft-lb

4. $BHP = \frac{\text{Torque} \times 2\pi \times RPM}{33{,}000} = \frac{2.93 \times 2\pi \times 2375}{33{,}000} = 1.32$ H.P.

5. $\text{Fuel lb/hr} = \text{cc's/sec} \times \frac{3600}{618\ cc/lb} = \frac{30\ cc}{178\ sec} \times \frac{3600}{618} = .982$ lb/hr

6. $\text{Fuel-lb/BHP-hr} = \frac{\text{Item 5}}{\text{Item 4}} = \frac{.982}{1.32} = .741$ lb/BHP-hr

Table XIII cont'd.

7. Fuel-lb/IHP-hr = $\frac{\text{Item 5}}{\text{Item 14}} = \frac{.982}{1.58}$ = .621 lb/IHP-hr

8. Brake thermal efficiency = $\frac{2545}{\text{Item 6 x 20,000}} = \frac{2545}{.741 \text{ x } 20{,}000}$ = 17.2%

9. Indicated thermal efficiency = $\frac{2545}{\text{Item 7 x 20,000}}$ =

$$\frac{2545}{.621 \times 20{,}000} = 20.5\%$$

10. Volumetric efficiency = $\frac{508}{RPM}\sqrt{\frac{h_1 - h_2}{w}}$ Item 7, Table XI Sample Calculations

$$= \frac{508}{2375}\sqrt{\frac{.783}{.0714}} = 71.1\%$$

11. Air fuel ratio = $\frac{\text{Item 15}}{\text{Item 5}} = \frac{13.15}{.982}$ = 13.4/1

12. Friction torque = (Zero reading - friction reading) x 2

= (.97 - .68) x 2 = .58 ft-lb

13. Friction horsepower = $\frac{\text{Torque x 2 x RPM}}{33{,}000} = \frac{.58 \text{ x 2 x } 2375}{33{,}000}$ = .262 H.P.

14. Indicated horsepower = Item 4 + Item 13

15. Air lb/hr = $.924\sqrt{w(h_1 - h_2)}$ lb/min x 60

$$= .924\sqrt{.0714(.783)} \times 60 = 13.15 \text{ lb/hr}$$

16. Mechanical efficiency = $\frac{\text{Item 4}}{\text{Item 14}} = \frac{1.32}{1.58}$ = 83.5%

Table XIV. Data in Tables XXVII, XXVIII, XXIX

Calculations same as for Table XIII

Table XV. Data in Table XXX

Calculations same as for Table XIII

Table XVI

1. Speed = 2200 RPM for example
2. Dynamometer reading = 160 lb data
3. Lever arm = 10.5 inches data
4. Torque = scale reading x lever arm/12

$$= 160 \times 10.5/12 = 140 \text{ ft-lb}$$

5. Brake horsepower $= \dfrac{\text{Torque} \times 2 \quad \times \text{RPM}}{33{,}000}$

$$= \frac{140 \times 2 \quad \times 2200}{33{,}000} = 58.7 \text{ H.P.}$$

Tables XVII, XVIII; XIX

Calculations same as for Table XVI

Table XX. Data in Table XXXII

1. Run No.1 for example
2. Speed = 2000 RPM data
3. Dynamometer scale = 115 lb data
4. Time/lb-fuel = 2' 32.5" data = 152.5 sec
5. lb-fuel/hr $= \dfrac{3600 \text{ sec}}{\text{Item } 4} = \dfrac{3600}{152.5} = 23.6$ lb/hr
6. Brake horsepower $= \dfrac{\text{Scale} \times \text{RPM}}{6000} = \dfrac{115 \times 2000}{6000} = 38.3$ H.P.
7. lb-fuel/HP-hr $= \dfrac{\text{Item } 5}{\text{Item } 6} = \dfrac{23.6}{38.3} = .615$ lb/HP-hr

Table XXI. Data in Table XXXIII

Calculations same as for Table XX

Table XXXI.

1. Tooth No.8 for example
2. Micrometer reading = .510 inches data
3. Zero reading = .609 inches data with valve closed
4. Valve lift = Zero reading - given reading

   $= .609 - .510 = .099$ inches

5. Crank angle = Tooth no. x 11.25^{o}/tooth

   $= 8 \times 11.25^{o} = 90^{o}$

6. Degrees per tooth = $\frac{360^{o}}{32 \text{ teeth}} = 11.25^{o}$/tooth